# VARIABLE SPEED CENTRIFUGAL COMPRESSOR

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**Abstract**

This paper describes a variable speed two stage 30HP centrifugal compressor developed for refrigerant R134a. Rotational speed of three phase AC induction compressor motor is regulated by varying the applied frequency of inverter. Two impellers are attached at both ends of the single shaft. The rotor section of motor is located in the middle of the shaft. The desired speed of the compressor motor is achieved via direct drive instead of gearbox. This makes the variable speed centrifugal compressor simple and compact.

Variable speed centrifugal compressor can change rotational speed when in operation. Thus capacity and pressure ratio can be adjusted by changing the drive frequency. This characteristics of the variable speed centrifugal compressor matches the real operating conditions of heat pumps and air conditioners. This can extend the operation range and reliability of space conditioning heat pump especially in a cold climate.

## 1. Introduction

Closed cycle vapor compression systems based on various refrigerants have been widely used in space conditioning, refrigeration and heat pump applications during the last century. These conventional systems will be widely accepted also in the early period of this century. Due to growing concern about global warming and air pollution problems, however, regulations for energy efficiency are required more strongly than ever. Deregulation of power industry throughout the world will make consumer pay higher rate for consuming electricity during peak electricity load period. Although alternative systems are actively under development, higher efficiency closed cycle vapor compression systems will be economical in the early 21 century. Research and development in improvement of motor, compressor and other components, and smart integration, and environmentally-friendly refrigerant and oil-free compressor will be active(ARTI 1997). Heat pumps in domestic, commercial and industrial application are considered as a feasible mean to reduce greenhouse gases. Although renewable energy systems are also available, closed cycle vapor compression heat pumps are the most economical in reducing $CO_2$ emission (Breembrock and Kleefkens 1999).

Conventional centrifugal compressors have been widely used in large scale chillers and refrigerators while positive displacement compressors are used in small and medium scale applications. Recently inverters are employed in positive displacement compressors, especially for small capacity applications, to improve efficiency and thus reduce energy consumption. This enables reduced transient on-off mode operation period and accurate temperature control and rapid cooling and heating. More than 30 percent reduction of energy consumption is reported in domestic air conditioners and heat pumps. Inverter-driven variable capacity compressors show advantage in multi-mode applications.

Rotational energy from constant speed motor is transferred to impellers via gearbox to increase rotor speed to the desired level in conventional centrifugal compressors. Rotational speed of motor in typical centrifugal compressor is constant at a low speed i.e. at 3,600 rpm. Gearbox is employed to increase the rotor speed to a higher level. This reduces the size of impellers and related components. Due to their complex structure and size, they are limited in large scale refrigeration.

Recent development of inverter technology and high speed bearing technology enables centrifugal compressors to rotate at higher speeds. Inverter direct drive of high rotational speed removes gearbox. And this high rotational speed reduces the size of components such as motor and impellers and shaft. Reduced size and removal of gearbox enable small and medium size centrifugal compressors affordable at lower prices. Inverter-driven high speed centrifugal compressors have reduced size and a smaller number of components and greater tolerance for manufacturing and assembly. This enables variable speed centrifugal compressors to be manufactured at a lower cost than positive displacement compressors and to be affordable at small and medium scale applications. Discharge pressure and flow rate change with varying rotational speed. With varied temperature conditions, variable speed centrifugal compressor can operate at appropriate pressure ratio without reducing flow rates. When the ambient air temperature falls with the same desired interior temperature, the pressure ratio needs to be reduced to maintain comfortable condition and reduce power consumption. Variable speed centrifugal compressors can save more energy than positive displacement compressors with maintaining appropriate pressure ratio.

Variable speed centrifugal compressors have variable capacity and variable pressure ratio characteristics. Inverters and controllers are also needed to be installed with them. Variable speed centrifugal compressors cannot be easily adopted in case of simply replacing constant speed postive displacement compressors. However, in case of needing energy efficiency and good quality temperature control, inverter driven positive displacement or centrifugal compressors are advantageous. As compared to variable speed positive displacement compressors, variable centrifugal compressors have advantages. They can control pressure and pressure ratio, and have good reliability for entering liquid refrigerant. With accurate pressure control, less frost is formed at heat exchanger surfaces. Liquid refrigerant can enter compressors inlet in defrosting cycle operation. With these characteristics, variable speed centrifugal compressors are advantageous in heat pump applications.

## 2.Experimental Procedure

Performance of variable centrifugal compressor is tested on two test rigs. One is based on ASME PTC 10 (ASME 1963). This rig is composed of compressors, condenser, expansion valves, and accumulators. Pressure transducers and thermocouples are installed at various location along refrigerant passage. Thermal stability can be achieved in a short period. Data that can be used in analyzing efficiency and related characteristics can be obtained conveniently. Based on measured pressures and temperatures and flow rates and input power, pressure ratios and adiabatic efficiency of impellers, motor loss and efficiency, and power consumptions, etc. are calculated to be used in updating design details. Figure 1 shows its installed view.

Figure 1. Test stand

Another test rig is used to measure compressors data more accurately. Water cooled condenser and evaporator and are incorporated in this test rig. Thermal stability is achieved in a longer period of time. However, compressor inlet temperature can be accurately adjusted. These two test rigs show a consistent agreement of their test results. In design stage, air cooled test rig was convenient to get data in a short period of time. However, water cooled test rig was accurate in maintaining desired temperatures.

## 3.Structure of Variable Speed Centrifugal Compressor

Variable speed centrifugal compressor is direct drive two-stage. The first and second stage impellers are attached at each end of shaft, while motor is located in the middle part of the compressor. Shaft rotor is composed of impellers, motor rotor, and shaft. Diffusers and shrouds are fixed at bracket bolted to casing. Motor stator lies in the middle of casing. Labyrinth seals and bearings are between impellers and motor stator.

Low pressure refrigerant gas enters compressor and high pressure refrigerant gas leaves compressor. Refrigerant gas enters into eye section of the first stage impeller. High speed gas, compressed and accelerated in the first stage impeller, leaves the impeller and enters into diffuser. In diffuser, a fraction of kinetic energy in high speed gas is converted to potential energy. Pressure rises and velocity decreases at the exit of diffuser. High pressure and low speed refrigerant gas is collected in the collector and then enters into the second stage.

A 30HP capacity compressor was developed. Refrigerant R134a was adopted. Designed cooling capacity of the compressor was 82,151 kcal/hr and expected heating capacity was 110,000 kcal/hr. In conceptual design stage, pressure ratio and flow rate were determined. Inlet and outlet conditions for each stage and motor input power were also estimated. Rotational speed of motor was designed at 24,000rpm. Saturation pressures based on condensation and evaporation temperatures 7.2°C and 54.4°C were 377.1 and 1,470 kPa and thus design total pressure ration was set to 3.9.

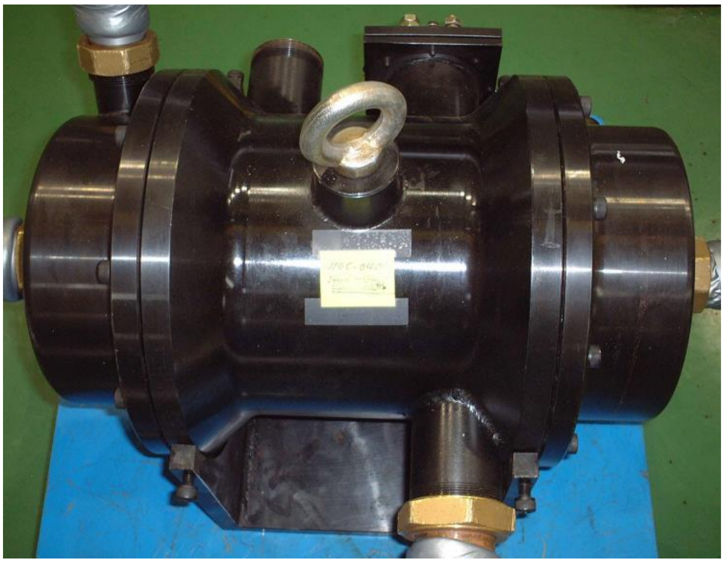


Figure 2. Two stage variable speed centrifugal compressor

3.1 Impellers and diffusers

Radial impellers are designed for R134a. The first stage and the second stage impellers may have different flow rates. Flash gas line connected between accumulator and inlet to the second stage impeller can make the second stage flow rate larger than the first stage flow rate in the two stage expansion cycle. A fraction of flow which is less than 10% of total flow rate can pass in the flash gas line. In one stage expansion cycle, the flow rate of each stage is identical. The current impellers need to be designed for identical flow rates.

| Stage | Temperature (°C) | Pressure (kPa) | Density ($kg/m^3$) | Speed of Sound (m/s) | Flow rate (kg/hr) |
|---|---|---|---|---|---|
| Stage 1 | 18.3 | 377.1 | 17.42 | 151.3 | 2332 |
| Stage 2 | 54.1 | 791.9 | 34.31 | 152.6 | 2332 |

Table 1. Inlet condition for each stage

Saturated pressure for 54.4°C is 1,470kPa. The resulting pressure ratios for the first and second stages are 2.10 and 1.86, respectively. The identical pressure ratios for the can be calculated as follows.

$$\Pi_I = \Pi_{II} = \sqrt{\Pi_C}$$

The identical pressure ratios make the axial force unbalanced. To make the axial unbalance smaller, the first stage pressure ratio becomes larger and the second stage pressure ratio becomes smaller. The designed outside diameters $D_2$ are 135mm and 121mm, respectively. To make the stage flow rate compatible, impeller tip heights $b_2$ are set to 5.9mm and 5.0mm.

3.2 Motor

Three phase induction motor was developed. Motor is cooled by the refrigerant gas before entering to the first stage impeller eye. Refrigerant gas channels are prepared around the stator. Cold refrigerant gas passes through the channel prepared at outside of the stator, and removes the heat from the stator. Refrigerant gas occupied between the rotor and stator is subject to viscous shearing action. The clearance between the rotor and the stator was determined to keep the windage loss as low as possible without losing a lot of efficiency. The density of refrigerant gas is high and the rotational speed is also high. Thus the windage loss becomes large. Outside diameter $D_R$ 68mm is and length $L_R$ is 110mm. Enthalpy gain of refrigerant gas from the inlet of motor to the exit of motor was measured to calculate motor losses, assuming that all of motor loss is converted to heat.

3.3 Bearing and seals

High speed ball bearings are employed. Recent development of high speed ball bearing technology made ball bearings possible to be adopted in high speed application like this. Axial load is much bigger than radial load. Bearing are mainly subject to axial load and care should be taken to accommodate axial load properly.

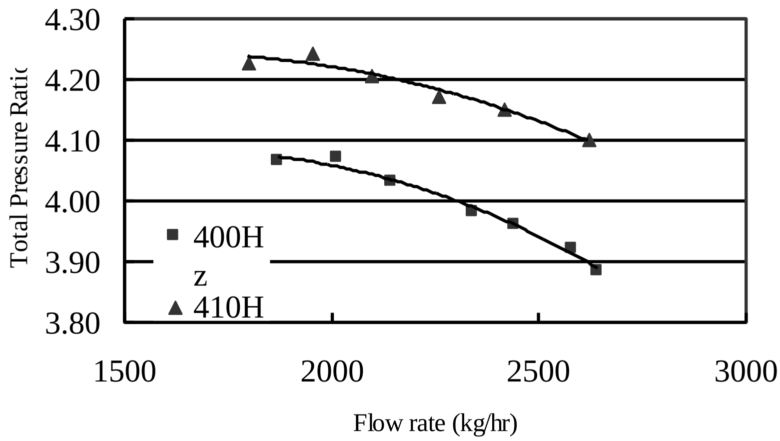
Recently, gas lubricated bearings are also under active development. Gas lubricated bearings have low load carrying capacity and low damping characteristics. High rotational speed is necessary for gas dynamic bearings. A rotational speed of 40,000rpm or higher is necessary and depends on gas bearing types and rotor weight. Despite of short history, foil bearings are considered to be employed for high speed turbomachinery applications.

Labyrinth seals are employed to minimize pressurized gas leakage between the rear side of impeller and the shaft. The number of land and land shapes were studied. Rotordynamic forces of labyrinth seal causes unstability.

**4.Test Results**

The ASME PTC 10 based test rig was used to acquire data during the development stage. Many sets of useful data could be obtained in a short time. Temperature and pressure transducers are installed at various locations. Flow sensors and powermeter are also installed.

Compressors have been tested at various flow rates. Rotational speed was also changed. Curves in Figure 3 display total pressure ratios $\Pi_C$ compared to varied mass flow rates. Pressure ratio generally decreases as the mass flow rate increases. When the rotational speed increases, flow rate and pressure ratio




increases.

Figure 3. Total pressure ratio vs. Mass flow rate

Motor loss is measured based on enthalpy of refrigerant gas passing the motor. Pressure and temperature of refrigerant gas are measured before the entry to the motor and after the exit of motor. As mentioned before, the windage loss is large compared to other components of motor losses. As shown in Figure 4, the total motor loss increases when the rotational speed increases.

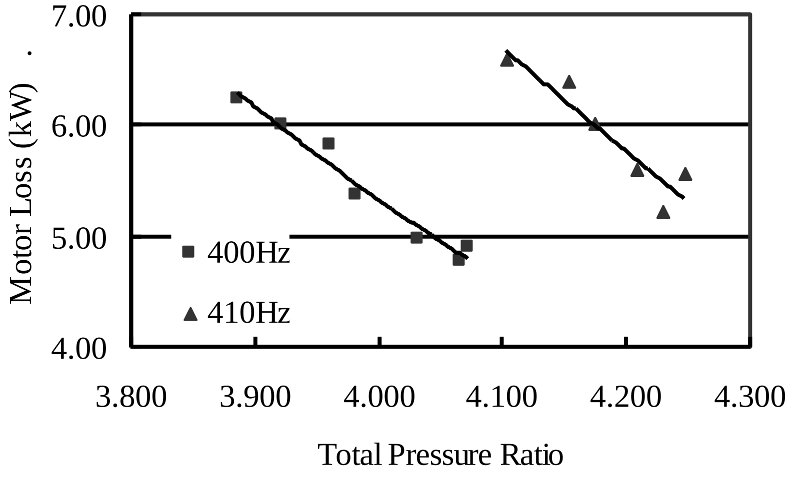



Figure 4. Motor Loss vs. Total Pressure Ratio

Cooling capacity is shown as the total pressure ratio varies. Cooling capacity decreases as total pressure ratio increases. It means that flow rate decrease with increased pressure ratio. Figure 5 show that pressure ratio increases with increased rotational speed. In this ASME PTC 10 based air cooled test rig, cooling capacity cannot be measured. Cooling capacity is calculated based on compressor inlet and outlet conditions.

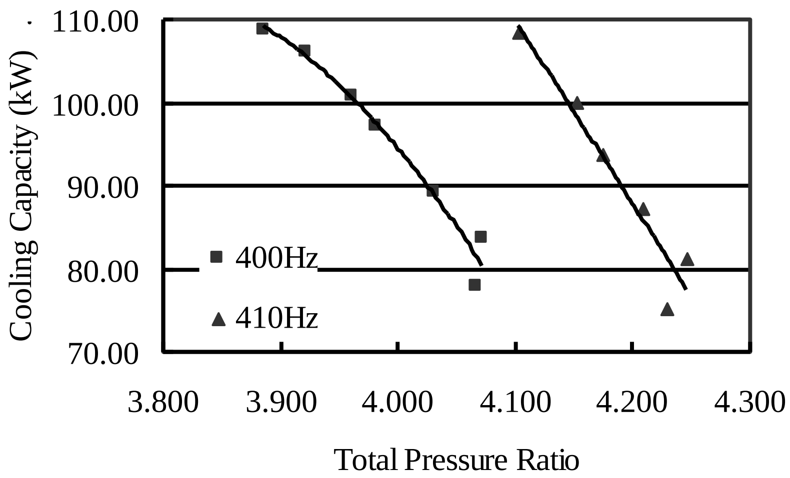



Figure 5. Cooling Capacity vs Total Pressure Ratio

The coefficients of performance for this compressor have been measured based on power consumption measurement. Powermeter is placed before the inverter. Power consumption of inverter varies, but about 1 kW is used in inverter when 30kW is consumed in compressor.

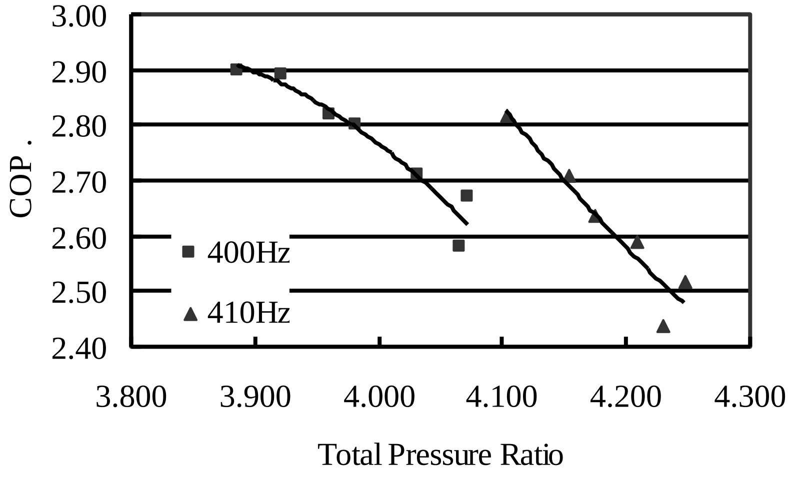



Figure 4. Coefficient of Performance vs Total Pressure Ratio

In water cooled compressor test stand installed at NAMRI, the compressor is tested. Shell and tube heat exchangers are used. Test results are listed in Table 2. They are in good agreement with ASME PTC 10 based test rig.

| | |
|---|---|
| Refrigerant | R134a |
| Cooling Capacity(kcal/hr) | 84609 |
| Consumed Power(kW) | 34.0 |
| Pressure Ratio | 3.896 |
| COP | 2.89 |

Table 2. Compressor Test Results on Water Based Test Rig

**5.Discussion**

Variable speed centrifugal compressors have good control over exit pressure and pressure ratio. Flow rates and pressure ratios increase with increased rotational motor speed. In pressure ratio vs. flow rate curves, high rotational speed curve move upward to the right. Low rotational speed curve lies in low pressure ratio and low flow rate zone.

Variable speed centrifugal compressors showed reliability for entering liquid refrigerant. Impellers do not contain refrigerant in an enclosed space during compression process. Refrigerant is accelerated by a centrifugal force in impellers. These variable speed centrifugal compressors are believed to be desirable for wet compression.

In space conditioning heat pump application, variable speed centrifugal compressors have desirable characteristics. First, variable speed centrifugal compressor based heat pumps operate not in a repeated start/stop manner but in a continuous manner. With proper control of fan with compressors, energy consumption can be reduced in a great deal. Second, temperature control quality is very good. Indoor temperature fluctuation is very small. Third, rapid heating and cooling can be achieved. Last, variable speed centrifugal compressor based heat pump can adjust pressure ratio during operation when the ambient temperature varies. Unnecessarily high pressure ratio operation can be avoided when low pressure ratio operation is necessary.

In hot water heat source heat pump applications, variable speed centrifugal compressors show desirable characteristics. When the hot water is used as heat source, compressors tend to be overheated. However, this centrifugal compressor controls exit pressure and temperature easily with controlling rotational speed, and can operate continuously within safe range.

Variable speed centrifugal compressors do not require any lubricating oil for contacting surfaces. This will facilitate adopting new refrigerant. High temperature refrigerants are necessary for industrial heat pump applications(IEA 1995). This centrifugal compressor is believed to be ideal for industrial heat pump applications.

## 6.Conclusion

Variable speed centrifugal compressor is developed and shows reasonable agreement with design objectives. It has variable pressure ratio and variable capacity characteristics. And it also have many desirable characteristics for heat pump applications. Its ability to control exit pressure and variable capacity characteristics show energy saving, and good quality temperature control.

Variable speed centrifugal compressor based heat pumps can contribute to reducing greenhouse gases and emission. High temperature refrigerant applications will be beneficial to reducing emission in industry.

**Acknowledgements**

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## Nomenclature

D = Outside diameter

b = Blade height

L = Length

Subscripts

I = the first stage

II = the second stage

C = Compressor

2 = tip of impeller
R= Rotor of motor
S= Stator

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